

Experimental characterization, modeling and simulation of a wood pellet micro-combined heat and power unit used as a heat source for a residential building

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ABSTRACT

Cogeneration provides heat and power in a more efficient way than separate production. Micro-cogeneration (micro-CHP) is an emerging solution for the improvement of energy and environmental assessments of residential buildings. A wood pellet Stirling engine micro-CHP unit has been studied in order to characterize its annual performance when integrated to a building. First, through a test bench experiment, both transient and steady state behaviors of the micro-CHP unit have been characterized and modeled. Then a more complete model representing a hot water and heating system including the micro-CHP unit and a stratified storage tank has been carried out. This model has been coupled to a building model. A sensitivity analysis by simulation shows that the dimensioning of different elements of the system strongly influences its global energy performance.

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1. Introduction

For a heat-to-work (or heat-to-electricity) conversion process, cogeneration designates the recovery of the residual heat to make it useful. Thus, cogeneration (or Combined Heat and Power, CHP) corresponds to the simultaneous generation of work (or electricity) and useful heat. Theoretically, this combined generation is more efficient than the separate generation of heat and power, implying lower primary energy consumption, and consequently a lower operating cost and lower environmental impacts. Large-scale cogeneration systems are widely used in process industries, commercial and public sector buildings, as well as for district heating [1]. Micro-CHP designates the low-powered CHP systems adapted to the heat and domestic hot water (DHW) provision for small residential and commercial buildings (nominal electric power below 10–20 kW according to sources) [1–3].

Due to small sizing, micro-CHP differs from larger CHP by the available conversion processes and by their efficiency. Nowadays, four main families of existing or emerging micro-CHP can be considered, differing by their conversion process: reciprocating engines, fuel cells, Stirling engines and steam engines (Rankine cycle)¹ [4]. Table 1, derived from Refs. [5–7], shows that energy sources and thermal and electric yields ranges differ for each

family. The Stirling engine micro-CHP systems seem to present several advantages for a use in residential buildings: low noise emission level, high heat recovery efficiency (60–90%) and electricity production efficiency range from 10% to 25% [8,9]. Moreover, the Stirling engine can operate with renewable fuels, such as liquid biofuels, biogas or wood, which may improve the environmental performance of the system, and consequently, of the building. Nevertheless, in 2009, still very few Stirling micro-CHP systems are commercialized. The instantaneous performance of Stirling engine micro-CHP systems is insufficiently known, especially those fueled by renewable fuels, like wood pellets. It may vary dynamically according to various parameters, as inlet water temperature and heating control and diffusion systems with which the micro-CHP unit is associated.

The objective of this study is to carry out one-year simulations of residential buildings heated by wood pellet-fuelled Stirling engine micro-CHP unit, in order to make the optimization of the performance of this innovative domestic heating device possible on an annual-mean approach.

In order to attain this objective, the performance and the dynamic behavior of a wood pellet micro-CHP unit have been characterized through a test bench. Then, from this characterization, a model has been developed and integrated to the model of a complete heating and DHW production device, coupled to a building model. The heating device includes the micro-CHP unit and a hot water storage tank. Finally, a sensitivity analysis has been carried out by simulation to understand the influence of key parameters on the mean annual performance of the system for residential applications.

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¹ Micro gas turbines CHP cannot be considered as micro-CHP yet because the nominal electric power of the smallest ones exceeds generally 30 kW.

Nomenclature

a	semi-major axis of the hyperbola (K)
b	semi-minor axis of the hyperbola (min)
C_{pw}	thermal capacity of water ($\text{J kg}^{-1} \text{K}^{-1}$)
LHV	lower heating value of the wood pellets (J kg^{-1})
HHV	higher heating value of the wood pellets (J kg^{-1})
L_{duct}	length of the duct (m)
M_{duct}	mass of water in the duct (kg)
m_{pellets}	mass of wood pellets (kg)
$\dot{m}_w$	mass flow rate of water (kg s^{-1})
p_{duct}	perimeter of the duct (m)
P_{el}	electric power (W)
$\bar{P}_{\text{pellets}}$	average higher heating power of the wood consumed (W)
P_{th}	recovered thermal power (W)
$\bar{T}_t$	mean temperature in the duct during the time step t (K)
T_{cw}	cold-water temperature (K)
T_{CC}	temperature inside the combustion chamber (K)
T_{ext}	outside air temperature (K)
T_{ref}	reference temperature in the start-up process (K)
t_{start}	duration of the START1 phase (min)
$T_{w,\text{in}}, T_{w,\text{out}}$	inlet and outlet water temperature (K)
T_{zone}	temperature in the building zone (K)
U_{duct}	overall heat transfer coefficient of the duct wall ($\text{W m}^2 \text{K}^{-1}$)
Δt	operating period duration (s)
ε	efficiency of a heat exchanger (–)
η_{th}	thermal efficiency (–)
η_{el}	electric efficiency (–)
φ_{cw}	phase of the cold-water function (rad)
φ_{ext}	phase of the outside air temperature fundamental (rad)
ω	annual angular frequency ($=2\pi/8760$) (rad h^{-1})

2. Literature review and modeling strategy

Very few studies have been carried out on Stirling micro-CHP systems yet, mainly due to the recent development of this technology. The modeling of the behavior of a micro-CHP based on a Stirling engine may follow various approaches, differing on the level of detail of the model.

An explicit modeling approach, focusing on the Stirling engine, involves the detailed modeling of the combustion process and thermodynamic cycle of the engine [10–13]. Such models aim at representing the dynamical behavior of the engine with very small

time steps. Moreover, their use requires a large amount of physical data of the engine, which may be difficult to get. This approach is over-detailed with respect to the requirements of the present work.

A more experimental study has led to the characterization of the steady state performance of the Stirling engine used for CHP [14]. It gives no details on the behavior of the system during transient modes. Nevertheless, this behavior has a strong influence on annual performance and must be taken into account in our model.

Another study has focused on the coupling of a Stirling micro-CHP unit to a heat storage and diffusion device, leading to the determination of the performance of the whole system when integrated to a building [15,16]. Apparently, this experimental study did not lead to the development of any dynamical model of behavior for the micro-CHP unit.

The annex 42 of IEA has followed a more global approach [17], offering a 3-component, grey box model, which has been integrated into three widespread dynamic simulation tools. In this model, four operating modes (stand-by, warm-up, normal and cool-down) can be distinguished and the modular structure allows to describe either internal combustion engine or Stirling engine micro-CHP units. Nonetheless, its time-step range is rather small (between one second to several minutes) and the calibration process leads to constant electric and thermal efficiencies, which is not adapted to the requirements of our study where a one-year simulation and the influence of the inlet water temperature on the performance are expected.

In this paper, the approach is more schematic, consisting in a simplified parametric model, based on the characterization data of both steady state and transient behaviors of the micro-CHP. A test bench experiment has been needed to characterize the micro-CHP unit behavior before developing the model.

3. Characterization of a micro-CHP unit

3.1. Tested device

A comparative study led by the authors, shows that, among about twenty micro-CHP devices already commercialized, only one runs on renewable fuel. This one, called “Sunmachine[®] Pellet”, is operated by wood pellets, for which the manufacturer announces a 3 kW electric power (Fig. 1). This device has been chosen for this study.

The Sunmachine[®] Pellet unit incorporates a 1-cylindre Stirling engine in alpha-configuration² providing a swept volume of 520 cm³. The working gas is nitrogen. According to the operating conditions, the working gas pressure may vary from 33 to 36 bar and the rotational speed of the Stirling engine from 500 to 1000 rpm. A condensing heat exchanger for the exhaust gas is included; thus, exhaust gas temperature stays well below 100 °C and compares to conventional condensing boilers. The different technical specifications given by the manufacturer of the micro-CHP unit are shown on Table 2.

The engine and generator of the Stirling-based CHP unit are manufactured as a single unit; therefore they have been studied and modeled as a single unit.

3.2. Experimental facilities and testing procedure

The test bench which has been designed for the characterization of the micro-CHP unit comprises two water circuits and is connected to the public electric grid so that the various energy and mass flows can be measured (Fig. 2). A closed water circuit

Table 1

Characteristics of the various types of micro-CHP devices, derived from Refs. [3–5].

Energy conversion device	Energy source	Conversion efficiency range (%)	
		Electric	Thermal
Internal combustion engine	Liquid fuel, natural gas	30–38	45–50
Fuel cell	Hydrogen, hydrocarbon	30–40	40
Stirling engine	Any type of fuel, solar radiation	10–35	60–90
Rankine cycle engine	Any type of fuel, solar radiation	10–20	70–85

² In this configuration, the two power pistons move in two separate cylinders [18].

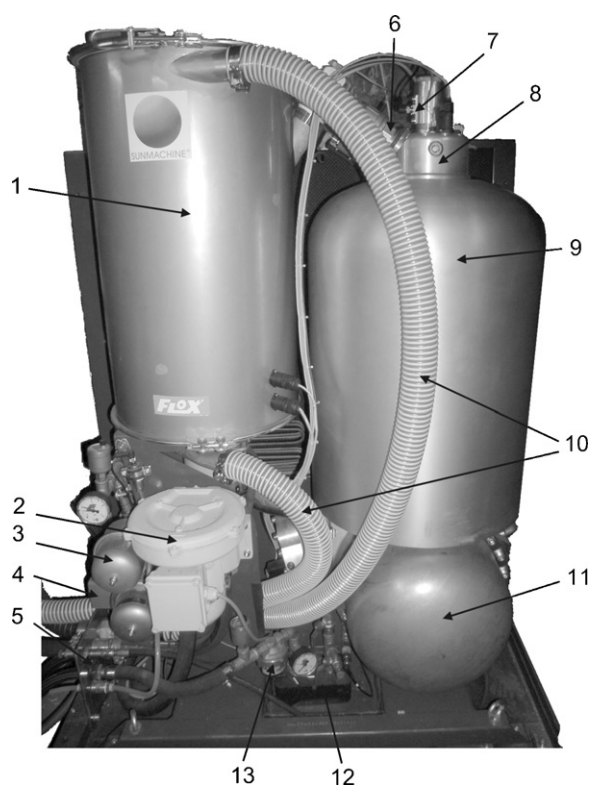


Fig. 1. Schematic illustration of the Sunmachine Pellet micro-CHP unit. 1: wood pellets storage tank; 2: ventilator for exhaust gases; 3: expansion bottle; 4: valve for pellets filling process; 5: connection support; 6: wood pellets feeder (endless screw); 7: lighting; 8: burner head and firebox; 9: combustion chamber; 10: pipe for pellets filling process; 11: Stirling engine and electric generator; 12: plate heat exchanger; and 13: secondary circuit water circulator.

simulates the domestic hot water production circuit. It includes a plate heat exchanger connected to a fresh water open circuit used for heat dissipation, electric heaters used for temperature control and a volumetric flow meter measuring the volumetric flow rate (VFR) of water. The temperature of water at the inlet and outlet of the micro-CHP unit as well as the fluid flow in the heating circuit

Table 2

Technical specifications of the tested micro-CHP unit (Sunmachine Pellet) given by the manufacturer.

Electric power output	~1.5–3 kW
Heat rate output	~4.5–10.5 kW
Electric efficiency (LHV)	~20–25%
Global efficiency (LHV)	90%
Maximum outlet temperature	70 °C
Optimal inlet temperature	<30 °C
Weight	~350 kg
Dimensions (width, length, height)	800/1200/1600 (mm)

can be recorded, which enables the calculation of the heat recovered by the water (1).

$$P_{th} = \dot{m}_w \times C_{p,w} \times (T_{w,out} - T_{w,in}) \quad (1)$$

The electric power output is measured and recorded directly by a wattmeter. The electric consumption of the different auxiliaries of the micro-CHP (pumps, fans, endless screw motor, inverter, monitoring and control panel) is measured separately. The circuit of the exhaust gases comprises a thermocouple, a volumetric flow meter and a gas analyzer to measure the temperature, the VFR and the composition of the exhaust gases.

3.3. Experimental results

The studies realized with the test bench aim at evaluating the performance of the micro-CHP in steady state over a wide range of inlet water temperatures and in two transient behaviors (Start-up and Shutdown). The temperature in the combustion chamber (T_{CC}) has been recorded during the transient evolutions because the internal controller of the micro-CHP unit uses this temperature to trigger the activation or the deactivation of several auxiliaries, which leads to the decomposition of each of these processes into several phases (Fig. 3).

3.3.1. Start-up process

The start-up process begins when the temperature measured in the storage tank (or simulated one) is 10 °C below the control temperature. It is divided into two different phases. The first one (START1) corresponds to the electric heating of the core of the combustion chamber. During this phase, T_{CC} varies following a

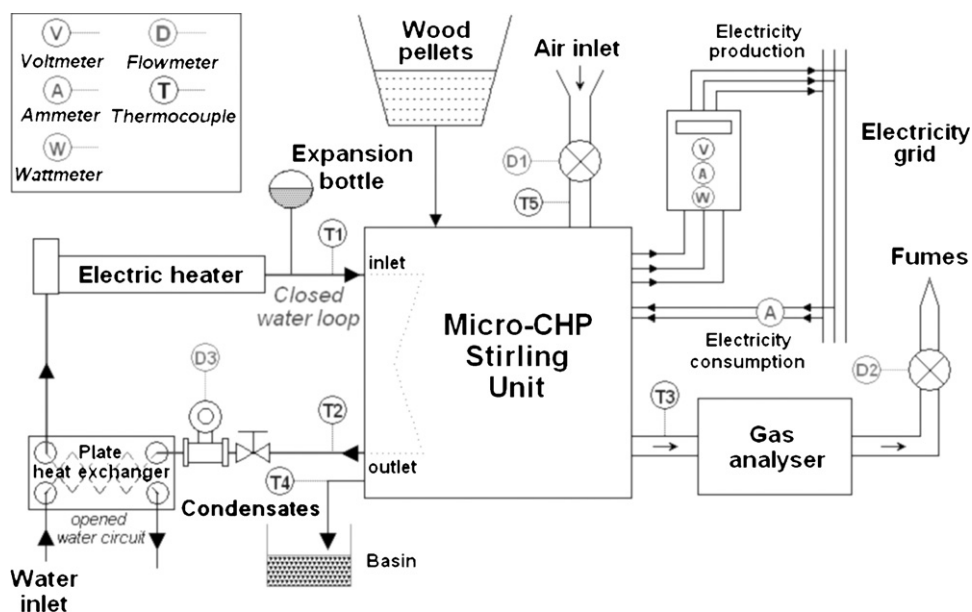


Fig. 2. Layout of the test bench.

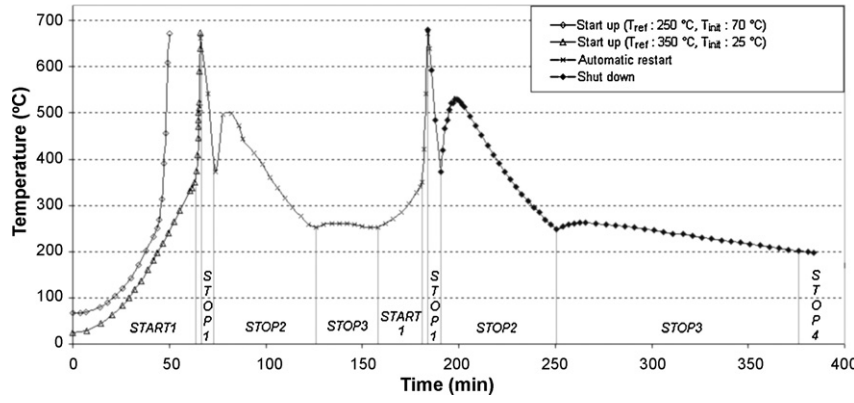


Fig. 3. Temperature measured in the combustion chamber for the three transient processes.

hyperbolic trend (Eq. (2)), where $a = 923$ K and $b = 68.7$ min. The triggering of the second phase occurs when T_{CC} reaches a reference temperature set by the operator (T_{ref} between 250 and 350 °C). Thus, the duration of START1, t_{start} , depends on both the initial and the reference temperatures (Eq. (3)); for a cold start-up it may vary from 43 to 63.3 min (see two first series on Fig. 3).

$$T_{cc}(t) = \sqrt{1 + \frac{t^2}{b^2}} \times a - (a - T_{cc}|_{t=0}) \quad (2)$$

$$t_{start} = b \times \sqrt{\frac{(T_{ref} + a - T_{cc}|_{t=0})^2}{a^2}} - 1 \quad (3)$$

Then, the combustion process begins (START2) until T_{CC} reaches 650 °C, which occurs after 3–6 min. This phase ends with the Stirling engine starting. During the whole start-up phase, the generator produces no electricity (the Stirling engine is shut down), but the power consumption of the auxiliaries is 2354 W during START1 and 220 W during START2.

3.3.2. Steady state operation

From the experimental tests, two major key parameters of the steady state performance have been identified: the cold-water temperature at the inlet of the CHP system and the heating load of the system. However, the CHP system is designed to operate only at full load. Therefore, a partial load is not possible and the only remaining key parameter is the inlet water temperature.

During the steady state operation, the instantaneous electric and thermal efficiency have been evaluated from the recorded data and Eqs. (4) and (5), according to various inlet water temperatures

(Figs. 4 and 5).

$$\eta_{el} = \frac{P_{el}}{P_{pellets}} \quad (4)$$

$$\eta_{th} = \frac{P_{th}}{P_{pellets}} \quad (5)$$

Two laws have been deduced from the data as piecewise linear functions (Eqs. (6) and (7)).

$$P_{el}(T_{w,in}) = \min(1380, 1600 - 6.5 \times T_{w,in}) \quad (6)$$

$$P_{th}(T_{w,in}) = \min(5400, 5850 - 17 \times T_{w,in}) \quad (7)$$

The correlation between these laws and the data cannot be very precise because of the dispersion of the recorded data, which seems to be due to the thermal mass of the gasification unit, to the combustion instability and to the heat recovery on exhaust gases. The adopted coefficients correspond to the average tendency of the performance. Here the maximum electric power output (1380 W) and thermal heat output (5400 W) were measured for inlet temperatures lower than 30 °C. In practice, inlet water comes from a hot water tank and most of the time its temperature is superior to 30 °C. These correlations show that the performance of the micro-CHP unit decreases while the inlet water temperature increases. Consequently, when coupled to a thermal storage tank, the performance of the micro-CHP unit will deteriorate when the temperature in the hot water storage tank increases.

Two different methods have been used to evaluate the mass of the consumed wood pellets.

The first one consists in running the micro-CHP unit for a relatively long period of time at a steady state operation regime.

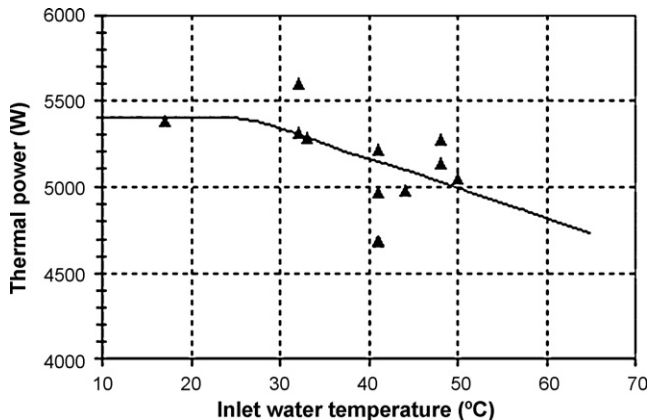


Fig. 4. Measured and modeled thermal performance of the micro-CHP unit.

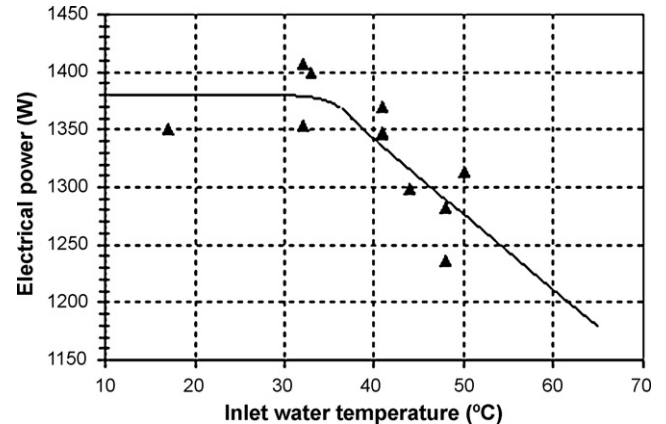


Fig. 5. Measured and modeled electric performance of the micro-CHP unit.

The mass of consumed wood pellets is evaluated by difference of level in the storage tank between the beginning and the end of this period. The energy delivered from the combustion of the wood pellets is calculated knowing the higher heating value (HHV) of the wood pellets. The average heat input power during this period is calculated by Eq. (8):

$$\overline{P}_{\text{pellets}} = \frac{m_{\text{pellets}} \times \text{HHV}}{\Delta t} \quad (8)$$

The second method consists in measuring the exhaust gas volumetric flow rate and its composition. The instantaneous mass flow rate of burnt wood pellet is calculated from the combustion equations of the wood pellets given in Refs. [19,20].

Following the first method, the measured value was 13.91 kg for 7.1 h of continuous operation (0.54 g s^{-1}). Following the second method, the instantaneous mass flow rate of burnt wood pellet is 0.45 g s^{-1} . The difference between the two values can be explained by incomplete combustion of wood, by the uncertainty in the measurements of wood pellet using the first method and by the fact that combustion in continuous operation continuously varies around a mean value. Considering a HHV of 19.3 MJ kg^{-1} for the wood pellets, the average heating power at the burner $\overline{P}_{\text{pellets}}$ is about 10.5 kW. Therefore, the maximum electric and thermal efficiency measured at an inlet water temperature of 30°C (best performance) are, respectively, 13.3% and 53.5%.

3.3.3. Shutdown process

The shutdown process is divided into four different phases. At first, the injection of the wood pellets is stopped, but the Stirling engine keeps running, the thermal power remains constant and the electricity power decreases with time (STOP1). This phase begins when $T_{w,\text{out}}$ is over a control temperature (here 80°C). The electricity produced during this phase is 85 Wh. After about 6 min, when T_{CC} is inferior to 400°C , the Stirling engine stops (STOP2). No more electricity is produced, but part of the heat stored by the thermal mass of the system is recovered from the engine and the exhaust gases. The thermal power output decreases at -16400 W h^{-1} during the first 13 min and -1100 W h^{-1} until it reaches 0 W. The electric energy consumed by the auxiliaries in this phase is 411 Wh. After 60 min, when T_{CC} decreases below 250°C , the exhaust gas fan is stopped (STOP3). The water circuit still evacuates heat. This phase lasts about 133 min and the electricity consumed by the auxiliaries during this phase is 557 Wh. When T_{CC} reaches 200°C , all water circulation pumps are stopped (STOP4). Thus the combustion chamber is cooled by conduction and natural convection. The temperature decreasing law is exponential with a time constant of 6.16 h. The electric consumption (22 W) corresponds to the consumption of the inverter. The annual consumption of the inverter is about

192 kWh, which is relatively high since it represents the production of 5.7 days of full load operation.

During the shutdown process, if the temperature measured in the storage tank (or simulated one) is 10°C lower than the control temperature, it might restart a start-up process (hot start-up). Actually this restart process only can occur when T_{CC} is below 400°C . Therefore, the hot start-up cannot occur less than 36 min after the shutdown process beginning. In this particular case, it will take about 23 min for the temperature of the combustion chamber to reach the 650°C necessary for the starting of the Stirling engine. The minimum duration between the beginning of the shutdown process and the restart of the Stirling engine is 59 min. It might take more time if the beginning of the hot start-up occurs while T_{CC} is already below 400°C . Such a hot start-up may occur several times during cold periods when the heating load is high.

4. Modeling of a hot water and heating system with a micro-CHP unit as a heat source

The energy assessment of the micro-CHP unit strongly depends on its dynamic behavior, which is linked to the correspondence between the micro-CHP unit dimensioning and the heating load it has to supply. Thus, this dynamic behavior has been studied when the micro-CHP unit is used as a heat source for a building. For this purpose, a model has been developed as a new module of COMFIE, software for the dynamic thermal simulation of buildings [21]. This model represents a complete heating and domestic water-heating device (Fig. 6), including the micro-CHP unit, a hot water storage tank and two heat exchangers for heating and hot water production. These four elements, linked by water ducts forming water loops, have been modeled using an object-oriented approach. Each element thermally interacts with the elements linked to it, but also with the building zone where it is installed. At each time step, given the hot water consumption (from a preset scenario) and heating loads (from a building model of COMFIE), the model computes electricity generation and consumption, and wood consumption. Each element of the model and their interactions are described hereunder.

4.1. Water loop

A water loop includes two active elements (amongst micro-CHP unit, heat exchanger, water storage tank and water consumption scenario), two water ducts and a water-circulating pump. At each time step, the water flow is calculated for every water loop, according to the corresponding control strategy (on–off control). A quasi-static behavior is assumed (no transient phases). For the micro-CHP unit water loop, the pump control is supposed to be linear with minimal and maximal flow rate. For the DHW water

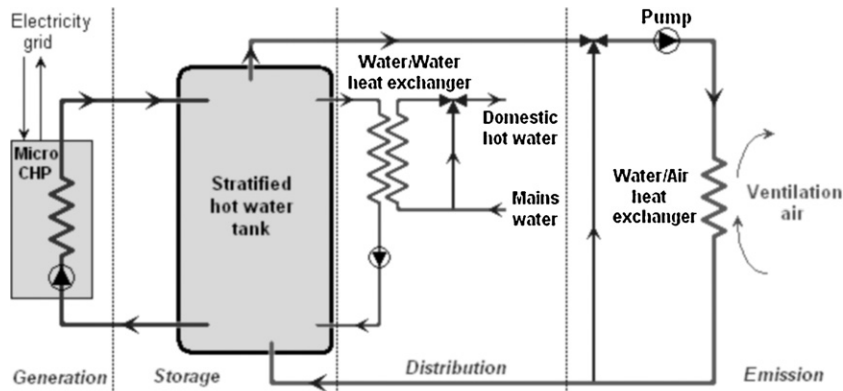


Fig. 6. Modeled air and domestic water heating device.

Table 3

Characterization of the micro-CHP unit working phases as modeled.

Working phase	Initial value of T_{CC}	Duration	Temperature evolution	Thermal production	Electricity generation	Electric consumption
STOP1	650 °C	6 min	−2500 °C/h	−6%/min (13 min)	850 W	374 W
STOP2A	400 °C	28 min	Variable	then −0.32%/min	0 W	374 W
STOP2B	400 °C	31 min	−280 °C/h	(68 min) till 0 W		
STOP3A	250 °C	45 min	Variable		0 W	66 W
STOP3B	250 °C	89 min	−33.7 °C/h	0 W		
STOP4	200 °C	Unspecified	Exponential	0 W	0 W	22 W
START1	Unspecified	t_{start} (Eq. (4))	Hyperbolic	0 W	0 W	2354 W
START2	T_{ref}	3 min	Linear	0 W	0 W	220 W
FUNC	650 °C	Until shutdown	Unspecified	See Eq. (8)	See Eq. (7)	220 W

loop and the heating water loop, the pumps control is an on–off type. Three-way valves are used to control the temperature of these two water loops.

4.2. Micro-CHP unit

The model of CHP unit is directly derived from the test bench characterization results. Two start-up phases, one operation phase and four shutdown phases, follow on from each other as described in Section 3.3. For a more precise description, the STOP2 and STOP3 phases have been divided into two sub-phases (A and B). The transition between a phase and the following one is controlled either by temperature or by timer. For each time step, the model computes the electricity consumption and the electricity and heat productions according to the current phase and the inlet water temperature (see Table 3). The temperature used to trigger the START1 phase is calculated at a preset height inside the stratified storage tank.

4.3. Stratified hot water tank

The model is derived from type 340 (Multiport store) from the TRNSYS library [22]. In this model, it is assumed that the stratified hot water tank is divided into horizontal, thermally uniform water layers. Each one exchanges heat by conduction through the tank walls, and by conduction and convection between the adjacent water layers. The equations are detailed in Ref. [23].

Water can be injected in or extracted from any water layer. Assuming a perfect stratification, hot water injection is supposed to take place in the layer with the closest temperature. In the present model, cold water is injected in the bottom layer of the tank and hot water is extracted from the top layer. The micro-CHP unit heats water taken from the bottom layer of the tank and injects it in the top layer.

4.4. Heat exchangers for heating

For the water-to-air and the water-to-water heat exchangers, a constant average efficiency ε is considered.

4.5. Domestic hot water production

DHW consumption flow and temperature are set as scenarios. Cold-water temperature may vary along the year. Thus the assumption of a constant cold-water temperature is unsuitable for an hourly dynamic simulation purpose and would lead to high inaccuracy. Very few dynamic cold-water temperature models have been developed [24–27] and none seems to be completely satisfactory yet. Most of these models represent the cold-water temperature T_{cw} as a one-year period sinusoidal function of time (Eq. (9)).

$$T_{cw}(t) = \overline{T_{cw}} + \Delta T_{cw} \times \sin(\omega \times t - \varphi_{cw}) \quad (9)$$

For each model, the yearly mean $\overline{T_{cw}}$, amplitude ΔT_{cw} and phase φ_{cw} are set from measured data or from correlations depending on several empirical parameters. Here, the model used is derived from Ref. [28], where $\overline{T_{cw}} = \overline{T_{ext}}$, $\Delta T_{cw} = \Delta T_{ext}/2$ and $\varphi_{cw} = \varphi_{ext}$.

4.6. Water duct

A water duct is modeled as a straight tube combined with a possible thermal insulation in contact with a building zone. Two distinct cases have been considered. When the water is flowing, the temperature varies exponentially with the considered position along the duct (10). When the water is stagnating, the temperature is homogeneous along the duct and thermal losses during a time step are calculated using (11). The heat exchanged through the duct wall is transmitted to the building zone where it is situated.

$$T_{w,out} = T_{zone} + (T_{w,in} - T_{zone}) \exp\left(-\frac{U_{duct} \times p_{duct}}{\dot{m}_w \times C_{pw}} \times L_{duct}\right) \quad (10)$$

$$\begin{aligned} \overline{T}|_{t+\Delta t} &= T_{zone} \\ &+ (\overline{T}|_t - T_{zone}) \exp\left(-\frac{U_{duct} \times L_{duct} \times p_{duct}}{M_{duct} \times C_{pw}} \times \Delta t\right) \end{aligned} \quad (11)$$

Thermal losses from each duct are transmitted to the building zone in which it is situated, which contributes to the reduction of the heating load.

5. Simulation and sensitivity analysis

The model described above, coupled to the building model of COMFIE, has been used to simulate the hot water and heating system with the micro-CHP unit equipping a residential building. It gives hourly data of the various outputs. The uncertainty due to the numerical convergence of the calculation has been evaluated and is negligible in comparison with the energy flowing through the system (generally inferior to 0.2%).

The reference simulation corresponds to a well-insulated house situated in France (heating load: 52 kWh m² year^{−1}, annual DWH load: 2605 kWh, treated floor area: 132 m²), in the oceanic climate zone (Fig. 7). Fig. 8 shows that the micro-CHP unit regularly refills the storage tank with heat and simultaneously produces electricity. The electricity generation efficiency actually decreases while the water temperature rises inside the tank. The electric consumption peak during each start-up procedures of the micro-CHP unit is compensated more or less after 1 h of electricity generation. In summer, when there is no heating load, the micro-CHP unit starts on every two days and operates continuously during only 2 or 3 h, which leads to a limited net electricity production. These results seem realistic but a complete validation of the whole model is necessary and will be carried out later, through a comparison with in situ experimental data.

A sensitivity analysis has been carried out for two main parameters: the annual heating load and the volume of the storage

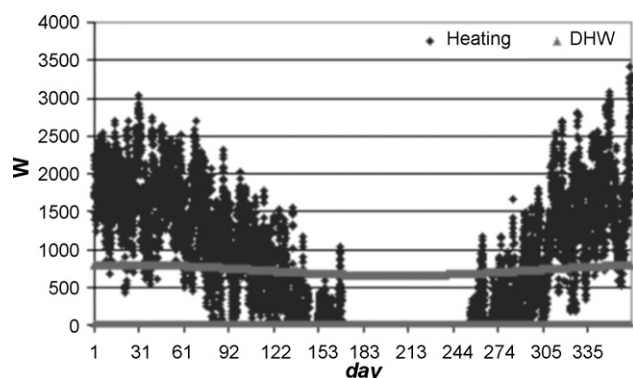


Fig. 7. Heating and DHW loads of the reference building.

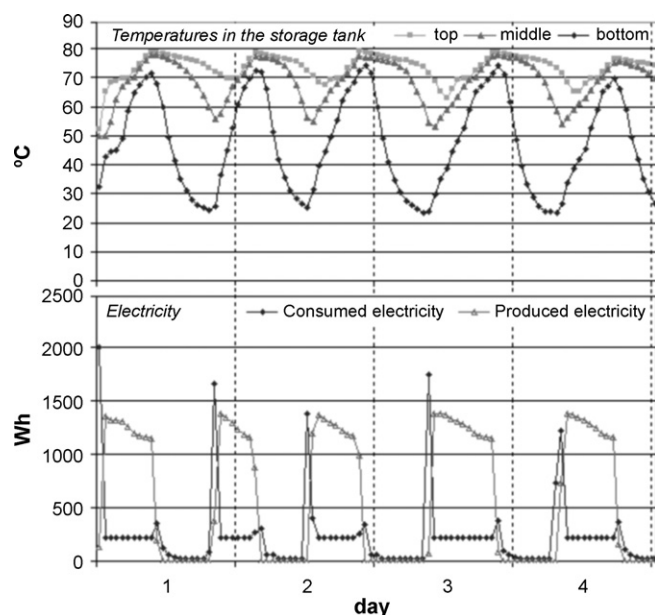


Fig. 8. Evolution of the temperature in three different water layers in the storage tank and of the power produced and consumed by the micro-CHP unit during four winter days.

Table 4

Annual performance of the whole system according to the heat loads (heating+DHW).

Heating+DHW loads (kWh)	Thermal losses (kWh)	Wood pellet consumption (kWh HHV)	Gross electricity generation (kWh)	Electric consumption (kWh)	Electric back-up (kWh)	Net electricity generation (kWh)
5707	1260	12 242	1579	1053	23	503
9508	1255	19 444	2519	1252	24	1243
10 828	1246	21 970	2869	1284	24	1561
12 583	1239	25 277	3317	1346	24	1947
16 394	1230	32 658	4300	1427	40	2833
18 382	1222	36 309	4833	1436	158	3239

Table 5

Annual performance of the whole system according to the storage tank volume.

Storage tank volume (l)	Heating+DHW loads (kWh)	Thermal losses (kWh)	Wood pellet consumption (kWh HHV)	Gross electricity generation (kWh)	Electric consumption (kWh)	Electric back-up (kWh)	Net electricity generation (kWh)
700	9508	1255	19 444	2519	1252	24	1244
1000	9507	1504	20 167	2621	1164	23	1435
2000	9502	2183	22 262	2883	955	21	1907
3000	9498	2764	23 778	3066	889	21	2156

tank. It shows that the electric performance of the whole heating system would be higher in case of higher heating loads (Table 4). Beyond 20 000 kWh of annual heating loads – which seems to be the optimum – the system is under-dimensioned and a back-up heater is needed, which deteriorates the global energy performance. In addition, a storage tank with a larger volume improves the net electricity generation due to longer and less frequent operating periods but it reduces the thermal performance, due to higher thermal losses through the storage tank wall (Table 5). To prevent the reduction of the heat recovery efficiency, a larger storage tank would have to be better insulated.

Some other simulations have shown that the thermal losses through the ducts wall or the bad control settings and dimensioning of the water pumps would reduce the global efficiency of the system.

6. Discussion

The model presented here has given interesting yearly results with short time calculation which fits the initial objectives of this study. Thanks to its object-oriented structure, this model may be used to optimize the dimensioning of each part of the system (micro-CHP, tank, pumps, etc.). Nevertheless, the comparison of simulation results to data taken from an in situ experiment is needed for its validation.

The performance of the micro-CHP unit emerging from the characterization and the simulations is inferior to what was announced by the manufacturer (Table 6). The first reason is that the tested CHP unit was a pre-series version with restrained power; this may also have influenced the various efficiencies. The problem remains in the global efficiency of the unit which is below what was announced. On the one hand, the heat recovery seems very efficient since the measured temperature of the exhaust flue gases has always been below 100 °C, but on the other hand a lot of uncertainties lie in the combustion process. The evaluation of the wood pellets consumption and of the efficiency of the combustion remains uncertain and could differ from what has been calculated.

Moreover, several defects have been noticed, that reduce the performance and should be corrected for the commercial version of the micro-CHP, like the unstable control strategy for the internal water pump and the repeated filling of the firebox grid with ashes. The quality of the combustion process should also be improved.

Table 6

Performance of the tested micro-CHP unit compared to the specifications given by the manufacturer.

	Tested micro-CHP	Manufacturer
Electric power output (kW)	≤1.38	~1.5–3
Heat rate output (kW)	≤5.4	~4.5–10.5
Electric efficiency (LHV) (%)	14.3	~20–25
Global efficiency (LHV) (%)	72.1	90

7. Conclusion

The dynamic behavior of a wood pellet micro-CHP unit has been modeled using data coming from a test bench. A numerical model has been realized which gives hourly data for such a unit supplying heat and hot water to a building. After a complete validation, the model could be used for the study of the optimal control of the different elements of the heating system.

The simulations show that the dimensioning of each part of the system (micro-CHP unit, storage tank, and also the ducts and heat exchangers) influences both the thermal and electric performances.

The best performance attained here by the whole heating system is not very high. It is mainly due to the limited performance of the micro-CHP unit, which does not match the manufacturer's specifications, and to the various losses (thermal losses of the hot water distribution and storage, auto-consumption of electricity) which represent about 7% of the heat and 32% of the electricity generated by the micro-CHP.

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